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SURGING OF 140 MW FRANCIS TURBINES AT HIGH LOAD, ANALYSIS AND SOLUTION

PULSATIONS À FORTE CHARGE SUR DES TURBINES FRANCIS DE 140 MW, ANALYSE ET REMÈDE

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Summary: High load surging is a specially delicate side of Francis turbines' dynamic behavior. There is yet no sure method to predict the occurrence of pulsations on a full-size machine, either from model test results or theoretical analysis.
Upon commissioning of the Saucelle-Huebra power plant, high load surging made it impossible to operate in the maximum power output range. The turbine designer brought the oscillations to an acceptable level by a modification of the runner cone, without air admission.
Thorough lab testing on a scale model allowed a better understanding of the problem. Influences of the runner cone, test head and tailrace tunnels are highlighted. A modal analysis using the cavitation compliance evaluated from model tests provides an explanation for the surge in the plant.

Résumé: La pulsation de forte charge est un aspect particulièrement délicat du comportement dynamique des turbines Francis. Aucune technique éprouvée ne permet encore de prédire l'apparition de pulsations sur une machine industrielle, à partir de calculs théoriques ou d'essais sur modèle réduit.
A la mise en service de la centrale de Saucelle-Huebra, les pulsations de forte charge empêchaient le fonctionnement à pleine puissance. Le concepteur de la turbine les a ramenées à un niveau acceptable par une modification de l'ogive de roue, sans admission d'air.
Une étude approfondie en laboratoire, sur modèle réduit, a permis de mieux cerner différents aspects du problème. On voit notamment l'influence de l'ogive, de la chute d'essai et des galeries de restitution. Une analyse modale de l'installation considérant la capacité de torche tirée des essais sur modèle réduit permet d'expliquer les importantes oscillations observées à la mise en service de la centrale.

Le texte en français peut être obtenu auprès des auteurs.

0. Introduction

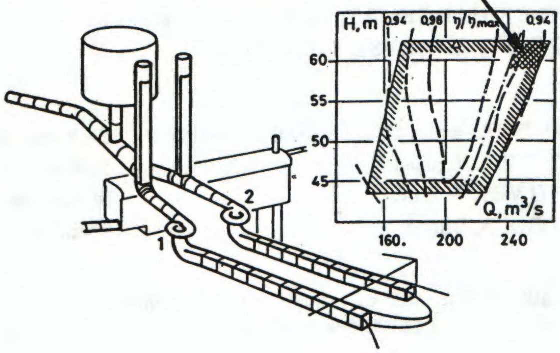
Prediction of Francis turbines' stability of operation [1, 4, 6] is among the major aims in present research on large hydraulic machines.

Investigations on pressure oscillations associated with Francis turbine operation made much progress over the past years [5, 6, 7, 13]. Several comparisons of full-size experiences with lab test results [2, 3, 8, 12] were attempted, yet no reliable technique is available for the prediction of prototype stability from theoretical considerations or from model tests. Preliminary stability analysis is seldom done. When a problem occurs, solutions [10, 17, 18] are drawn from a set of methods. However, chance sometimes has an important part in their results.

In this paper, we describe a full-load instability which made it impossible to operate the plant in its maximum power output range. The designer's solution to the problem is described. A lab test program gave a better insight to essential aspects of the problem. We show that an appropriate model test procedure [13] could have helped in anticipating the full-size turbine's stability problem. Concepts for the transposition of model data to prototype predictions are presented. Key elements are highlighted and restrictions are discussed.

1. Saucelle-Huebra: description of the power plant and field tests

Figure 1: Schematic view of the plant layout and range of operation of the turbines with unstable region.



At the junction of rios Duero and Huebra, in the Salamanca province of Spain, the Saucelle-Huebra power plant is part of the Duero hydro-electric project. A 1330 m long headrace tunnel takes the water from the dam to the surge tank. From there, a branching penstock with gate shafts feeds the plant's two nq = 82 high specific speed Francis turbines. Runners are 5.23 m in diameter with a rated power of 137 MW at the maximum head, 62 m (608 J/kg). The cavitation parameter under 62 m is $\sigma = 0.28$, reference elevation being the runner outlet. Draft tubes are connected to the tailwater through tailrace tunnels 109 and 118 m long.

Operation is guaranteed beyond rated gate opening, up to 140 MW in the high head range (figure 1).

Upon commissioning of the plant, surging occurred above 130 MW for high heads with the lowest tailwater. The air admission device (through the shaft seal and runner cone) did not allow natural air entrainment. Although the pulsation was acceptable with higher tailwater levels, operation of the units was temporarily restricted to 128 MW.

Modifications were undertaken on unit 2 to help free air admission through the runner cone: a funnel-shaped snorkel pipe (figure 4) was fitted on the cone so as to reach the level where the vortex cavity was observed in model tests. Instruments were set up to record pressure and power output fluctuations. Compressors were installed for a wider exploration of air admission possibilities.

With the air valve shut, we went through the maximum power output range, up to 140 MW. A high load pulsation is clearly visible in the test results. Its amplitude is much lower than at commissioning. Unit 1 (without the snorkel pipe) still produced unacceptable oscillations in this range of operation.

Figure 3 shows the power output fluctuations of unit 2 at 138 and 139 MW. If unit 2 runs alone, oscillations are acceptable. If unit 1 is also set at high load, the dominant frequency is shifted and annoying pulsations from unit 1 are felt all the way to unit 2.

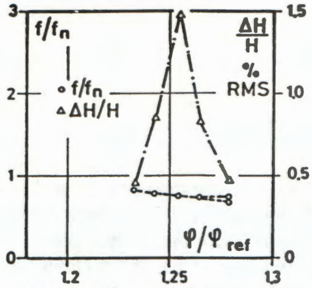


Figure 2: High load pressure pulsations on the prototype draft tube wall, unit 2

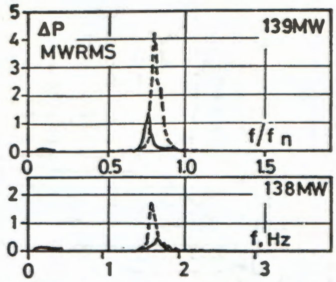


Figure 3: Power output pulsations at unit 2.
—— unit 2 alone, – – – unit 2 + unit 1

Additional testing with small forced air admission flows lowered oscillation frequencies and amplitudes. As just the presence of the snorkel pipe was enough to quiet the high load surge, air admission was abandoned. The runner cone of unit 1 was modified as in unit 2. Since then, operation of the plant is unrestricted in the whole guaranteed range, without loss of generating efficiency.

2. Lab tests on a scale model

Even though the owner was fully satisfied to see the instability problem solved in such a simple way, some questions still wanted answering.

- Why wasn't the full load instability detected during the initial model tests in the designer's lab?
- How can the snorkel pipe quiet the surge without drawing air?
- How do the long tailrace tunnels affect the oscillations?

The HYDRO VEVEY turbine model was erected on IMHEF's test rig. Its dynamic behavior was closely investigated according to a test procedure elaborated in these last years [13]. The model runner is 0.28 m in diameter. The main test head is 15 m. This is chosen for best performance of the test installation and measurement systems.

On all diagrams in this paper, pressure signals are read from a sensor on the draft tube wall, on the inner side of the elbow (figure 4). Pressures were also measured on other locations in the model. A comparison of different pressure signals is mandatory for a diagnosis of the machine's dynamic behavior. The present analysis is primarily based on pressure signals from sensors around the draft tube cone and at the spiral case inlet, as well as on torque fluctuations.

2.1 General survey

Figure 4 shows the amplitude spectra of pressure signals at the draft tube wall. Frequencies are relative to the rotational frequency and amplitudes, to the test head. The test is run at best efficiency specific energy ψ_{ref} with specific flows φ ranging from part load ($0.5\ \varphi_{ref}$) to full load ($1.33\ \varphi_{ref}$). This is done for three runner cone configurations:

- the sharp cone as used during the initial model tests in the designer's lab,
- the cutoff cone as in the prototype original design,
- the cutoff cone with the funnel-shaped snorkel pipe as on the modified unit.

Draft tube vortex cavities are plotted as seen through the model's Plexiglass draft tube cone.

Part load oscillations are basically the same with all three runner cones, but important differences are visible in the full load region. In all cases, a well-structured pulsation develops from $1.2\ \varphi_{ref}$ up to full gate. The maximum amplitude at $1.3\ \varphi_{ref}$ is $\Delta H/H = 1.2$ % RMS with the sharp cone and 2.45 % RMS with the cutoff cone, but goes down to 0.85 % RMS with the snorkel pipe. Full load pulsations may develop over very small variations of gate opening; measurements must be taken at closely spaced guide vane angles.

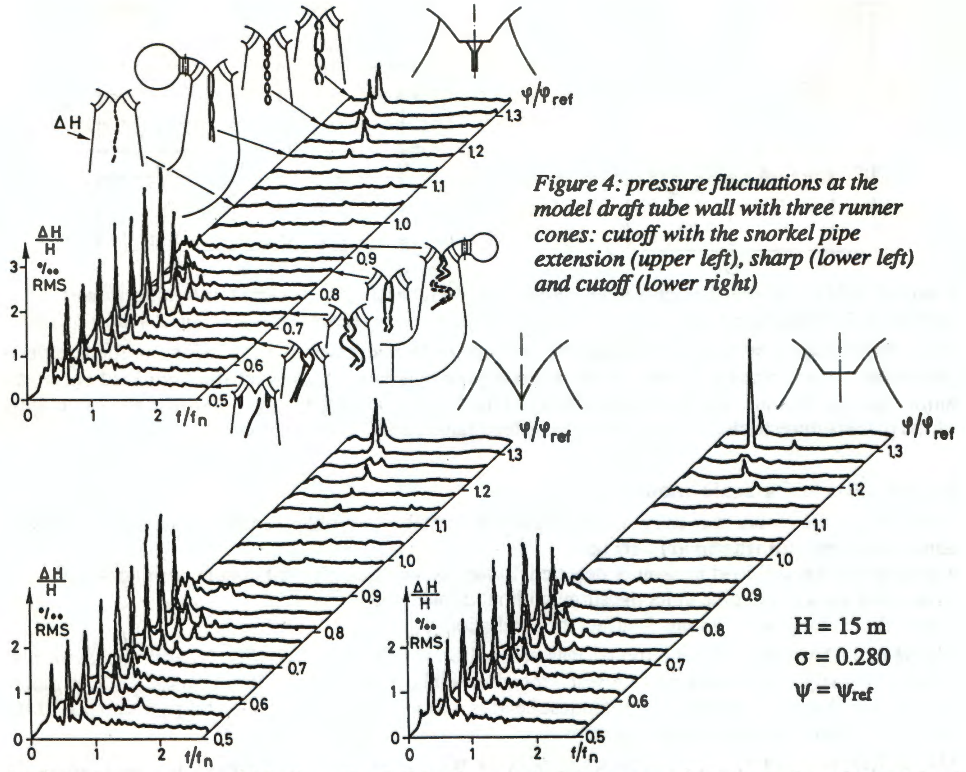
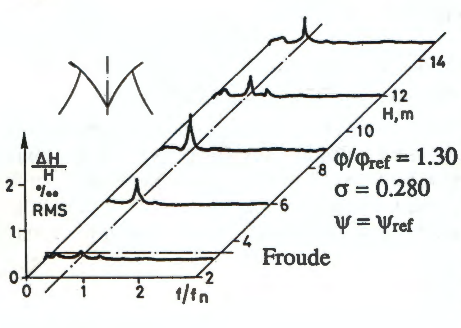


Figure 4: pressure fluctuations at the model draft tube wall with three runner cones: cutoff with the snorkel pipe extension (upper left), sharp (lower left) and cutoff (lower right)

At first glance, one could believe the full load surge to be no more dangerous than the part load fluctuations. However, the full load surge is a global oscillation of the water column. With equal amplitudes, it will communicate to the feed pipes oscillations much greater than the rotating pressure field associated with the part load precession [9, 11, 19]. The full load surge may endanger stability of operation as soon as its periodicity is established, provided that it can excite a characteristic frequency of the hydraulic system or the generator's subsynchronous oscillations.

2.2 Influence of test head and model-prototype comparisons

Figure 5: Influence of test head on the full load surge.



Additional tests showed that a well-organized pulsation always occurred at full load, around ψ_{ref}, in wide ranges of test head and cavitation parameter σ. The amplitude was systematically greater with the cutoff runner cone than with the sharp one.

The test head doesn't change the full-load pulsation's relative frequency and has little influence on its auto-correlation. These known rules [12] are illustrated on figure 5 at 1.3 φ_{ref}, $\sigma = 0.28$ with the sharp runner cone, down to Froude's head, 3.3 m. Relative amplitudes have small variations in the covered domain. The signal is ill-defined at the lower test head, due to the instruments' limited dynamic range.

Whatever the test head, a fair pulsation develops at 0.63 fn in these operating conditions. The test head has no influence on the draft tube vortex cavity volume, and changes in the appearance of the cavity are minor.

The influence of the snorkel pipe extending from the cutoff runner cone was closely examined. Relative frequencies and amplitudes of the full load surge for three test heads are shown on figure 6, with the cutoff cone and with the "stabilizing" snorkel pipe. Variations of the maximum amplitude and of the corresponding flow-rate don't seem very coherent, but two observations stand out of this test:

- The test head and snorkel pipe have little influence on the relative frequency of oscillations.
- The snorkel pipe's moderating effect is felt only under 15 m of head. Under 10 m, oscillations are even stronger with the snorkel pipe.

With the same scales of relative flow, frequency and amplitude, figure 7 shows the pressure fluctuations at the draft tube wall of the model and prototype turbines operating in homologous conditions. In both cases, a pulsation ridge develops at frequencies lower than the rotational frequency. Relative frequencies are the same for model and prototype. Maximum amplitudes and corresponding flow-rate are different: $\Delta H/H = 1.5$ % RMS at $1.255\ \varphi_{ref}$ for the prototype against 1 % RMS at $1.31\ \varphi_{ref}$ for the model.

Model tests clearly show the danger of full-load surge. Relative frequencies are the same as on the prototype. However, neither the relative amplitudes nor the flow-rate above which pulsations will become unacceptable may be predicted from these results. The model tests did not systematically show a stabilizing influence of the snorkel pipe.

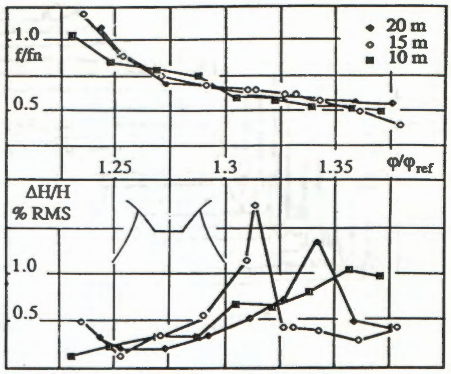


Figure 6: Evolution of the full load surge at three test heads, without and with the snorkel pipe. $\sigma = 0.28$, $\psi = \psi_{ref}$

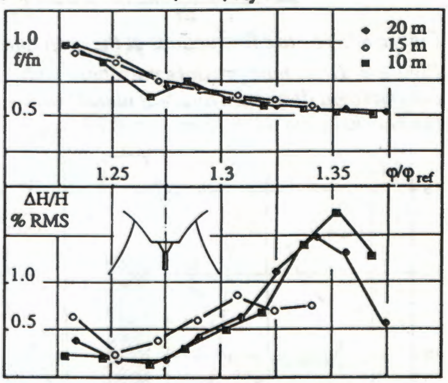
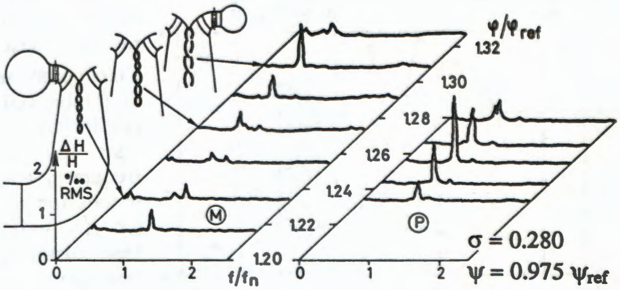


Figure 7: Full-load pressure oscillations on the model (M) and prototype (P) draft tube wall.



2.3. Influence of the tailrace tunnel

All these model tests were performed with a conventional draft tube ("0/2", figure 8), opening in a downstream tank where a free water level ensured a zero impedance boundary. The Saucelle-Huebra turbines are connected to the tailwater through long tailrace tunnels. To investigate the influence of these additional channels on the machines' dynamic behavior, we also tested the model with two lengths of tailrace tunnels, "1/2" and "2/2". Although this doesn't achieve similitude of dynamic coupling, the longer is geometrically similar to the one in unit 2 of the real-life installation.

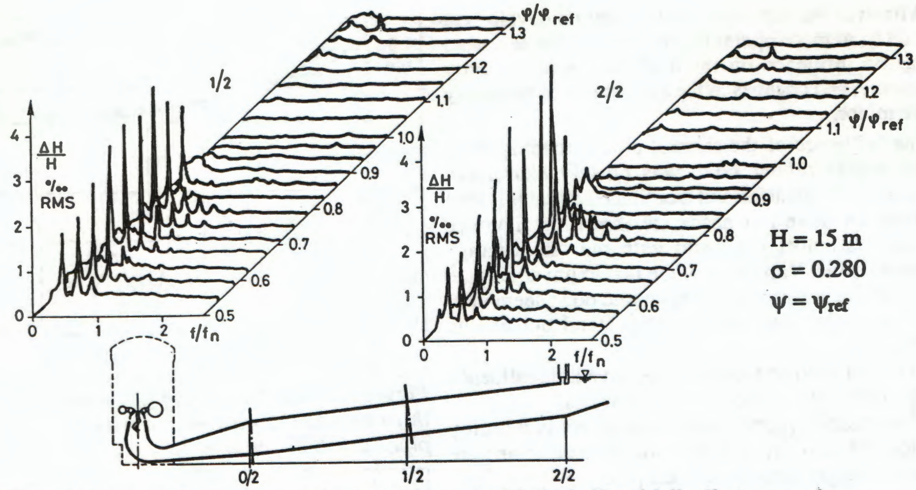


Figure 8: Pressure fluctuations at the draft tube wall with the half and full tailrace tunnels.

Figure 9: Draft tube pressure pulsations with conventional draft tube and half tunnel, with the snorkel pipe. $H = 15$ m, $\sigma = 0.28$, ψ_{ref}

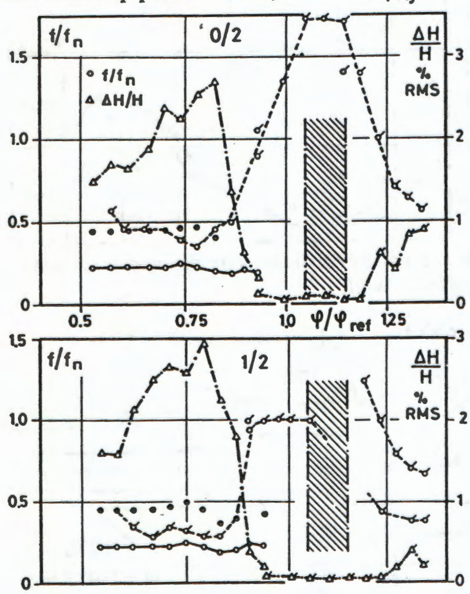


Figure 8 shows the amplitude spectra of draft tube pressure oscillations, from part load to full load, with the half and full tailrace tunnels. As compared to results with the conventional draft tube (figure 4), the half tunnel hardly modifies part load pressure oscillations. With the full tunnel, resonance is experienced at 0.8 φ_{ref}. The lengths of tailrace tunnel strongly affect the full load surge. The pulsation ridge is still visible below rotational frequency from 1.2 φ_{ref} to 1.3 φ_{ref}, but the base frequency of free oscillations is brought down to lower values. At 1.3 φ_{ref}, $f_0/fn = 0.65$ with the conventional draft tube against $f_0/fn = 0.375$ with the half tunnel. Frequencies get even lower with the full tunnel. Amplitudes are subdued, and tracing the evolution of the free oscillation in the full tunnel test is rather uneasy.

Chief components of the pressure oscillations at the draft tube wall are plotted as functions of the flow-rate for the conventional draft tube and half tunnel tests. (o) stands for the part load precession frequency and (Δ) for its amplitude, (•) is the precession's second harmonic frequency and (σ, Δ) represent free oscillations and the full load pulsation. The hatched strip indicates vortex-free operation. The precession relative frequencies and amplitudes are nearly identical in both tests. The lower free oscillation frequency is systematically smaller with the half tunnel. This is roughly explained by the influence of the added masses (the additional draft tube length) on the resonance frequency of the tailrace water passages seen as a simple oscillator.

3. Analysis of test results

The full load pulsation [12, 15, 16] is basically a free oscillation of the system made up of the water plug in the draft tube and the vortex cavity's compliant volume. This oscillation is generally excited only by the wide band hydraulic noise generated within the flow. In some cases, a flow instability produces a low amplitude, narrow band excitation. If the frequency of this disturbance happens to fall within a resonance frequency band, amplitudes may become excessive.

It was shown that the model runner outlet admittance is negligible when seen from the draft tube. The upstream boundary limit in lab tests is then quite simple. If we assume all compliance in the draft tube to be located at the runner outlet in the vortex cavity, the frequency f_0 of free oscillations varies like the rotational frequency fn if operating conditions are in similitude. If λ_x is the ratio of (x) values for homologous geometries and flow patterns,

$$f_0 = \frac{1}{2\pi}(C\,L)^{-1/2};\quad C = -\frac{1}{E}\frac{\partial Vol}{\partial \sigma};\quad L = \int \frac{dl}{A} \text{ and } \lambda_{f0}^{-2} = \lambda_C\,\lambda_L = \lambda_R^3/\lambda_E \cdot 1/\lambda_R = \lambda_R^2/\lambda_E = \lambda_{fn}^{-2} \text{ if } \lambda_\psi = 1$$

Incidentally, the frequency of a periodic disturbance generated by flow instability should obey Strouhal's similitude law: $\lambda_{fe} = \lambda_Q/\lambda_R^3 = \lambda_{fn}$ if $\lambda_\varphi = 1$

Relative frequencies of both periodic disturbances and free oscillations can then be approximated to transpose from one test head to another. The disturbing frequency is transposable from model to prototype. Free oscillation frequencies would be too, if similitude of boundary conditions could be achieved.

This leads to the following explanation for what we observed in model tests and field measurements:

- Beyond the vortex-free region, a slight flow instability occurs in the runner outlet central region. This is felt through a low amplitude oscillation. Replacing the sharp runner cone with the cutoff cone strengthens the instability and excitation.
- The excitation frequency is a little bit lower than the rotational frequency. All things otherwise equal, it goes down when the flow-rate increases.
- The test head has a small influence on the excitation relative amplitude. This accounts for the maximum amplitude and associated flow-rate being dependent of the test head.
- The snorkel pipe doesn't really have a stabilizing effect. It slightly modifies flow patterns in its immediate vicinity. That way, amplitudes may be significantly altered in some operating conditions.

Compliance of the water column inside the draft tube is actually not negligible. At best, it may be included in the lumped cavitation compliance, but even this is questionable. For a smarter evaluation of draft tube compliance lumped at the runner outlet, we did as follows: The mean wave propagation speed in the draft tube is identified from the draft tube geometry and the frequency of free oscillations observed on the model running in the vortex-free region. For each operating condition, we then compute the hydraulic impedance at the runner outlet for the measured free oscillations frequency. It is then easy to figure the capacitance required to tune the system. If it exists, it stands for the cavitation compliance, lumped at the runner outlet.

Figure 10 displays the results of this analysis as a function of flow-rate. This is done with the conventional draft tube and the half tailrace tunnel. The cavitation compliance is in a non-dimensional form: $C^* = C \cdot E/R^3$ [12, 13, 19]. It is quite coherent with the observed cavity volumes (figure 4). Cavitation compliances should of course be independent from draft tube length. The results may be assumed to verify this.

The model tailrace tunnel is a thin-walled prismatic duct. The wave propagation speed may then be expected to be very low. Measurements showed it to be close to S = 100 m/s. In these conditions, the free oscillation has harmonics in the considered frequency band. This explains the presence of several pulsation ridges in the full load range [12, 16].

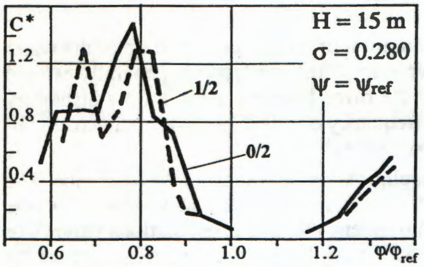


Figure 10: Estimation of the cavitation compliance with the conventional draft tube and half tailrace tunnel

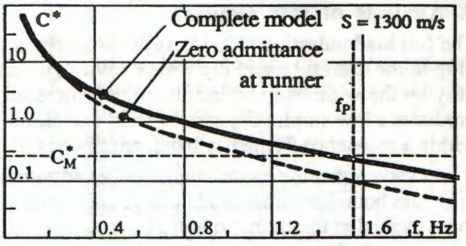


Figure 11: Non-dimensional cavitation compliance and free oscillation frequency. f_P: frequency from prototype data; C^*_M: non-dimensional cavitation compliance estimated from model tests.

It would naturally be possible to compute in the same way a reduced wave propagation speed in a draft tube portion arbitrarily assumed to be influenced by the presence of the vortex cavity. This would make a finer behavior model, specially in the part load range where vapor structures may extend past the draft tube elbow. We feel, however, that the present lumped approximation is sufficient in this case.

4. Stability of the prototype piping system

The non-dimensional cavitation compliance $C^* = C \cdot E/R^3$ represents all additional compressibility in the draft tube, lumped at the runner outlet, due to turbine operation in given conditions of specific flow, energy and cavitation parameter σ.

Figure 11 shows as a function of frequency the non-dimensional cavitation compliance required to tune the free oscillations in unit 2 of the prototype layout, unit 1 being shut down. Two computations based on hydraulic impedance theory are run with different assumptions for the upstream boundary condition. In one case, the runner admittance is set to zero, as seen in model tests [11, 12]. In the other case, the runner outlet impedance is computed from the water intake, surge tank, gate shafts and shut unit 1 limits, using a reliable behavior model [14] for transmission through the turbine.

Pulsation of unit 2 alone occurred at 1.56 Hz (0.75 f_n) for operation at 258 m³/s (1.255 φ_{ref}) with $\sigma = 0.28$. Model tests indicated a non-dimensional cavitation compliance $C^* = 0.26$ in these conditions. These values are plotted on figure 11. The non-dimensional cavitation compliance for free oscillations at 1.56 Hz is fairly close to the one evaluated from model tests. Wave propagation speed in the tailrace tunnel was set at 1300 m/s for this calculation. This may seem much, but it qualifies water under a 1.5 bar pressure in a fully rigid duct. All vapor generated by turbine operation is lumped into C^*. Also, we saw during the field tests that the water leaving the tailrace tunnels was completely free of bubbles.

Simplified calculations with a full reflection at the runner don't let reasonable wave propagation speeds account for the oscillations observed on the prototype. To compute the full-size circuit's response, the upstream piping's dynamics can't be overlooked.

Modal analysis is done on the case of simultaneous operation of both units at full load. Pulsations were observed at 1.68 Hz (figure 3). Overlooking changes in σ, we assume the non-dimensional cavitation compliance at unit 2 to remain $C^*_2 = 0.26$. We first seek unit one's cavitation compliance C^*_1 so that oscillations may tune at 1.68 Hz. C^*_1 will be complex. If its real part is positive, we drop the imaginary part and compute the frequency and damping of free oscillations in the system. Finally, the distribution of pressure and flow-rate amplitudes over the whole piping system is computed. Magnitudes are set to align with oscillations observed on unit 2. We do find a possible system oscillation at 1.68 Hz. The non-dimensional cavitation compliance at unit 1 is then $C^*_1 = 0.19$.

According to model test data, this is met at $\varphi/\varphi_{ref} = 1.24$. Unit 1 flow indeed was a little bit smaller than unit 2's.

This analysis was run for several C_2^* values. Free oscillations always settled at 1.68 Hz with small variations of C_1^* and damping. The piping system makes resonance at this frequency possible in a wide range of cavitation compliances.

The distribution of amplitudes associated with the free oscillation (figure 12) allows commenting upon the influence of various circuit elements on system dynamics. Note the break in flow-rate distribution at the surge tank insertion. The headrace tunnel's dynamics can't be overlooked. The mass storage effect of the gate shafts calls for special attention. These elements are often forgotten in stability analyses. Here, they have a great influence. We can even wonder whether the whole problem could not have been solved simply by raising the gates to reduce the throttling effect in the shafts.

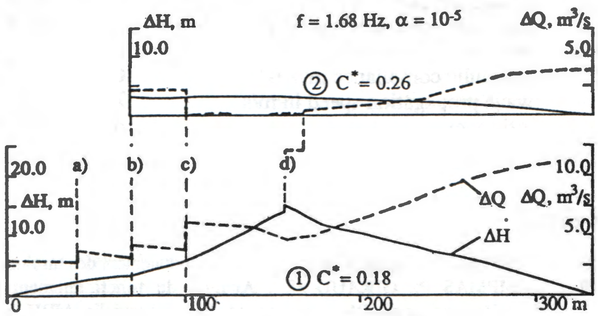


Figure 12: Representation of mode shapes in the plant layout at 1.68 Hz. a) surge tank insertion, b) manifold, c) gate shafts, d) units.

5. Conclusions

Upon commissioning of the Saucelle-Huebra power plant, full-load oscillations impeded operation in the maximum power output region. They were brought down to an acceptable level by a minor modification of the runner cones. All restrictions to operation in the guaranteed domain were removed without any loss of efficiency.

Lab tests on a scale model according to IMHEF's procedure clearly showed the potentiality of full load pulsations. However, the quieting influence of the runner cone modification that had such a spectacular effect on the prototype wasn't systematically found. The exact action of the snorkel pipe wasn't fully explained.

The runner and draft tube geometries must be exactly reproduced for proper lab testing of a turbine's dynamic behavior. The similitude of seemingly minor elements such as the runner cone is essential.

Tests must be run all over the potentially unstable ranges of operation. Observed relative frequencies may be compared with characteristic frequencies of the plant. In the case of full load oscillations, more attention should be paid to the pressure signal auto-correlation than to its amplitude. A well-established full-load oscillation in model tests may not even be a nuisance on the prototype.

The dynamic influence of long tailrace tunnels can't be simulated in model tests. Such a feature requires separate calculations. The non-dimensional cavitation compliance, a key parameter for these calculations, may be estimated from model tests with a conventional draft tube.

In the case of the Saucelle-Huebra plant, calculations show that the full-load disturbance excited the piping system's fundamental mode. This explains the strong surging experienced upon commissioning of the plant.

List of symbols

φ	specific flow-rate: $\varphi = Q/\pi R^3\omega$	ψ	specific energy: $\psi = 2E/R^2\omega^2$
$..._{ref}$	at best efficiency point taken as reference	σ	Thoma's cavitation parameter
Q	flow-rate in m^3/s	E	hydraulic energy per unit mass in J/kg
P	power in MW	H	head in m
R	runner outlet outer radius in m	ω	angular speed in s^{-1}
fn	rotational frequency in Hz: $fn = \omega/2\pi$	f	frequency in Hz
ΔQ	amplitude, flow-rate fluctuations	ΔH	amplitude, pressure fluctuations
ΔP	amplitude, power fluctuations	RMS	root mean square amplitude (standard dev.)
A	pipe cross-sectional area in m^2	l	length in m
Vol	draft tube vortex cavity volume in m^3	L	hydraulic inertance in m^{-1}
C	hydraulic compliance in $m \cdot s^2$	C^*	non-dimensional cavitation compliance
S	wave propagation speed in m/s	λ	scale factor
η	efficiency	α	damping coefficient

Sources

[1] CAMPMAS P. : Stabilité du régime de fonctionnement des turbines Francis. AIRH Symposium Nice (1960)

[2] CAMPMAS P., GIRAUD H. : Analyse du fonctionnement d'une turbine Francis: comparaison des investigations sur modèle réduit et sur turbine industrielle. AIRH Symposium Nice (1960)

[3] CASACCI S., WEGNER M., HENRY P., PRENAT J.E. : Examen expérimental de la stabilité des turbines Francis sur modèle et sur prototype à charge partielle. IAHR Symposium Amsterdam (1982)

[4] DERIAZ P. : A contribution to the understanding of flow in draft tubes of Francis turbines. IAHR Symposium Nice (1960)

[5] DÖRFLER P. : Evaluation concepts for low frequency oscillations in hydraulic reaction turbines. IAHR W.G., Milan (1983)

[6] DÖRFLER P. : Francis turbine surge prediction and prevention. Waterpower '85, Las Vegas (1985)

[7] FANELLI M.: Research on off-design behaviour of Francis turbines: an overview of present state, difficulties, open problems, needs and strategies. IAHR W.G. Milan (1991)

[8] FISHER R.K., ULITH P.: Comparison of draft tube surging of homologous scale models and prototype Francis turbines. Voith Research and Construction vol 28e (1982)

[9] GRAESER J.E., PRENAT J.E., WEGNER M.: Report of experiments on a Francis turbine and its test loop. IAHR W.G. Milan (1983)

[10] GREIN H. : Vibration phenomena in Francis turbines: their causes and prevention. IAHR Symposium Tokyo (1980)

[11] HENRY P., GRAESER J.E., PRENAT J.E., WEGNER M.: Scale effects concerning hydraulic quasi-stationary oscillations on a turbine model and its test circuit. IAHR Symposium Stirling (1984)

[12] JACOB T., MARIA D., PRENAT J.-E.: Comportement dynamique d'une turbine Francis à forte charge. Comparaisons modèle - prototype. SHF Comité technique 134, Paris (1987)

[13] JACOB T., PRENAT J.E., GRENIER R.: A characterization procedure for the dynamic behavior of Francis turbines: Practical comparison of elbow and Moody type draft tubes. IAHR Symposium Trondheim (1988)

[14] JACOB T., PRENAT J.E. : Identification of a hydraulic turbomachine's hydro-acoustic transmission parameters. IAHR W.G. Milan (1991)

[15] PRENAT J.E., JACOB T. : Investigating the behaviour at high load of a Francis turbine model. IAHR W.G., Mexico (1985)

[16] PRENAT J.-E., JACOB T.: Comportement à forte charge d'un modèle de turbine Francis et effets d'échelle. AIRH Symposium Montréal (1986)

[17] THICKE R.H. : Methods of controlling turbine draft tube vibrations and stability Canadian electrical association, hydraulic power section (1980)

[18] VATCHER T.R., HUNTER M., PERREAULT G., COULSON D.M. : Hydraulic resonance in a 78 MW Francis turbine - field test results. IAHR Symposium Strirling (1984)

[19] WEGNER M. : Cavitation as erosion and instability cause - experiments and remedies. Lahore (1983)

[20] WYLIE E.B., STREETER V.L. : Fluid transients. éditions FEB Press, Ann Arbor (1985)